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# MODELING OF COMPRESSOR VIBRATION FOR IMPROVED DYNAMIC DESIGN

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## ABSTRACT

The two main sources contributing to scroll compressor vibration at running frequency are the residual rotational unbalance in the crank-rotor assembly and the reciprocating force due to the Oldham ring. The intent of this paper is to develop a rigid body model of the compressor vibrating on resilient grommets. This model is a design tool capable of identifying the sensitivity, of the steady-state vibration levels, to manufacturing process tolerances of counterweight and crank-rotor parts. Other capabilities of the model include prediction of transient start-up vibration thus identifying key parameters for its control. Also using this model one can predict any potential resonances that could occur with new compressor design supported on standard grommets; this enables us to modify the existing grommet designs to tune away the resonances and also provide the vibration isolation desired.

## INTRODUCTION

The two main mechanical components contributing to the scroll compressor vibration are the residual rotational unbalance in the crank-rotor assembly and the reciprocating force due to the Oldham ring. It has been observed that the compressor vibration at the running frequency is primarily a rigid body mode. Hence the intent of this analysis is to develop a simple model based on the concept of a rigid body (entire compressor) vibrating on resilient supports or grommets. In order to simplify the analysis symmetry is assumed leading to the uncoupling of some of the modes of vibration. The periodic excitation sources for this model are the residual rotating unbalance which can be obtained from measurements on a balancing machine or from an estimation of the process tolerances in the assembly of the crank-rotor and counterweight parts, as well as a residual reciprocating force due to the Oldham ring.

## VIBRATION MODEL

Figure 1a shows a schematic of the compressor mounted on grommets and Figure 1b shows the relevant dimensions used in the vibration model as well as the grommet shear ( $K_S$ ) and compression stiffnesses ( $K_V$ ). The governing equations for a rigid body on resilient supports can be derived from using a Lagrangian approach and is given below in matrix differential equation form (neglecting damping) as:

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{F} \quad (1)$$

$$\mathbf{M} = \text{diag}[m_c \quad m_c \quad m_c \quad I_x \quad I_y \quad I_z]; \quad \mathbf{K} = \sum_{j=1}^4 \mathbf{\Gamma}_j^T \mathbf{S} \mathbf{\Gamma}_j; \quad \mathbf{u} = \{u_x \quad u_y \quad u_z \quad \theta_x \quad \theta_y \quad \theta_z\}^T \quad (2a-c)$$

$$\mathbf{\Gamma}_j = \begin{bmatrix} 1 & 0 & 0 & 0 & -L_G & -r_f \sin(\alpha_j) \\ 0 & 1 & 0 & L_G & 0 & r_f \cos(\alpha_j) \\ 0 & 0 & 1 & r_f \sin(\alpha_j) & -r_f \cos(\alpha_j) & 0 \end{bmatrix}; \quad \mathbf{S} = \begin{bmatrix} K_S & 0 & 0 \\ 0 & K_S & 0 \\ 0 & 0 & K_V \end{bmatrix}; \quad \alpha_j = \alpha_0 + (j-1)\frac{\pi}{2} \quad (3a-c)$$

Here  $\ddot{\mathbf{u}} = \frac{d^2 \mathbf{u}}{dt^2}$ ,  $\mathbf{M}$  is the mass matrix,  $\mathbf{K}$  is the stiffness matrix,  $\Gamma_j$  is the co-ordinate transformation matrix relating the center of gravity location of the compressor to the various feet locations,  $m_c$  is the mass of the compressor and  $I_x, I_y, I_z$  are the mass moments of inertia about the X, Y and Z axes (with the origin at the center of gravity of the compressor) respectively,  $L_G$  is the vertical distance between the center of gravity and the feet and  $\alpha_0$  is the angle between the X axis and location of foot 1 (see Figure 1b). In the next section the various terms of the force vector  $\mathbf{F}$  on the right hand side of equation (1) are developed.

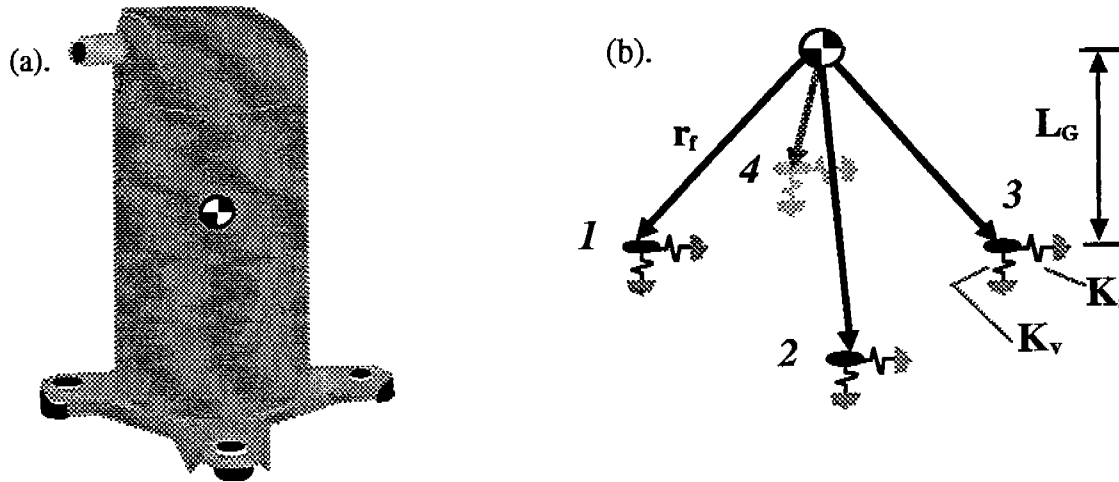


Figure 1. Schematic of compressor on grommets and dimensions used in the vibration model.

#### Force And Moment Excitations

As mentioned in the introduction the main sources of excitation at the running frequency are the residual rotating unbalances in the crank-rotor and counterweight assembly and the residual reciprocating force of the Oldham ring. Figure 2a shows the angular locations of the various forces and Figure 2b shows the vertical locations of these forces. The equations for the various forces and moments are given below where  $\omega$  is the running frequency in rad/s and  $\theta = \omega t$  where  $t$  is the time in seconds and  $R_{or}$  is the orbit radius:

$$\mathbf{F}(\theta) = \mathbf{F}_U(\theta) + \mathbf{F}_L(\theta) + \mathbf{F}_{Old}(R_{or}; \theta) \quad (4)$$

$$\mathbf{M}(\theta) = \mathbf{L}_U \times \mathbf{F}_U(\theta) + \mathbf{L}_L \times \mathbf{F}_L(\theta) + \mathbf{L}_{Old} \times \mathbf{F}_{Old}(R_{or}; \theta) \quad (5)$$

#### CASE STUDIES

In order to verify the model, measurements of the running vibration mode shapes were compared to that predicted by the model and the comparison can be seen in Figure 3; although the entire motion is an ellipse only one half of it is plotted as the shape is symmetric. The comparison between experiment and theory is pretty reasonable as can be seen from Figure 3. The inward dips seen in the experimental mode shape is due to the presence of asymmetry in an actual compressor, due to suction and discharge tubing.

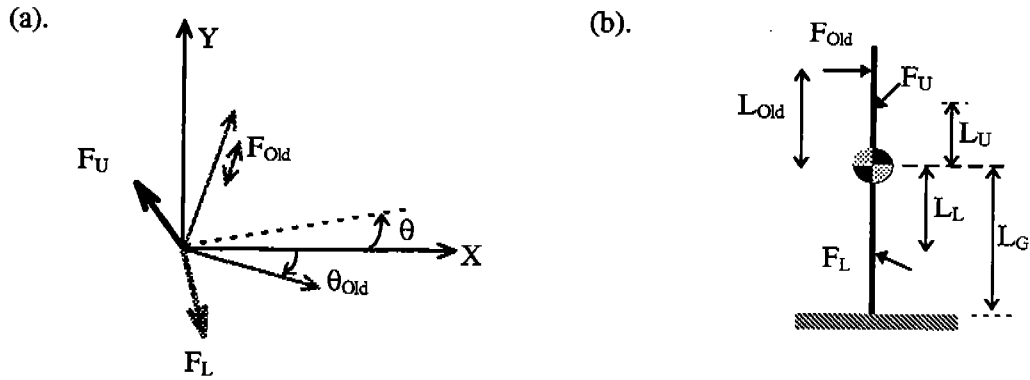


Figure 2. Angular and vertical location of the various forces exciting a scroll compressor.

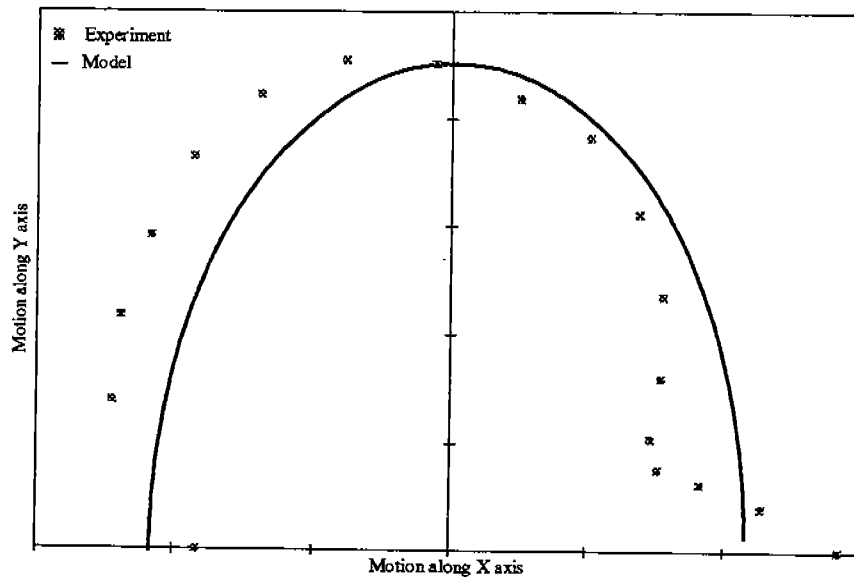


Figure 3. Comparison between predicted and measured vibration mode shapes.

Having confirmed the validity of this model, the next step is to use this as a tool to estimate the effect of production process tolerances on the vibration level variations. Two typical effects examined here are the tolerances on the masses of the counterweights and those on the angular misalignment of the counterweight center of gravity relative to the crankpin flat. Figure 4 shows the impact of the deviation of the counterweight center of gravity angular location on the vibration levels; it also shows comparison with experimental measurements. From this figure it is clear that the model does reasonably well in predicting the effect of the angular misalignment. Figure 5 shows the effect of counterweight mass variation on the overall vibration levels; only model prediction is included here. From these two figures it is clear that vibration levels are more sensitive to the angular tolerance than the mass variation in the counterweights.

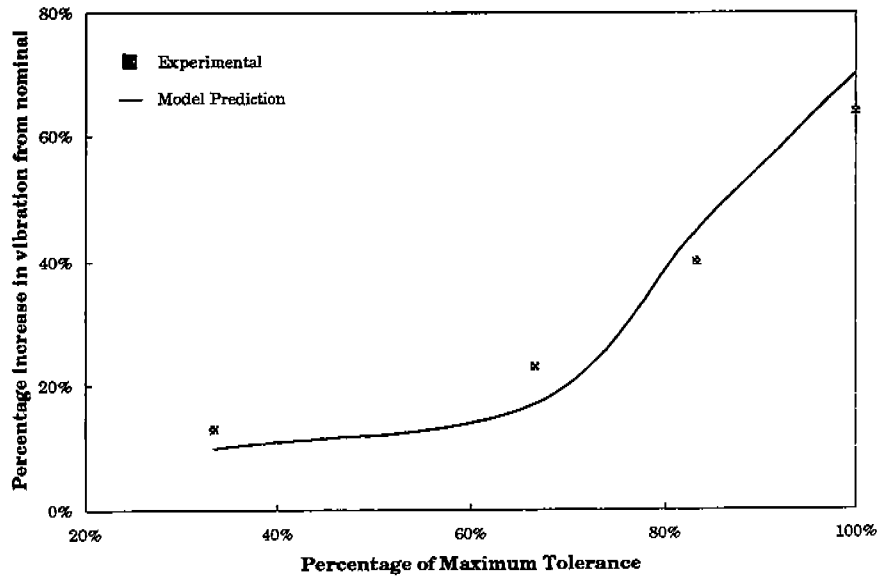


Figure 4. Effect of counterweight angular alignment tolerance on vibration levels.

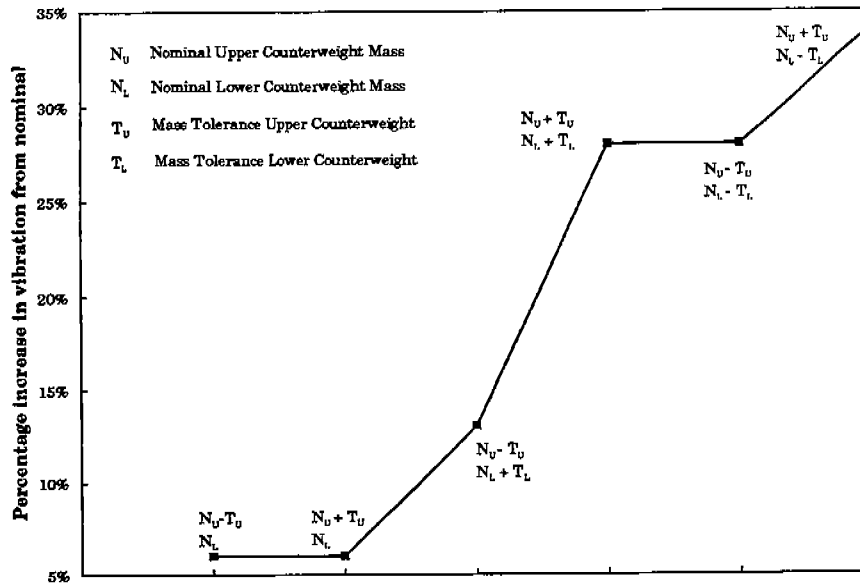


Figure 5. Effect of counterweight mass tolerances on vibration levels.

## CONCLUSION

In this paper a rigid body model for scroll compressor vibration has been developed to identify key parameters controlling vibration levels. One can predict any potential resonances that could occur with a new compressor design using a database of current grommet stiffnesses as a function of pre-loads. Further desired grommet compression and shear stiffnesses can be calculated to provide optimal vibration isolation. This model is proposed to be used for the prediction of transient start-up vibration levels.